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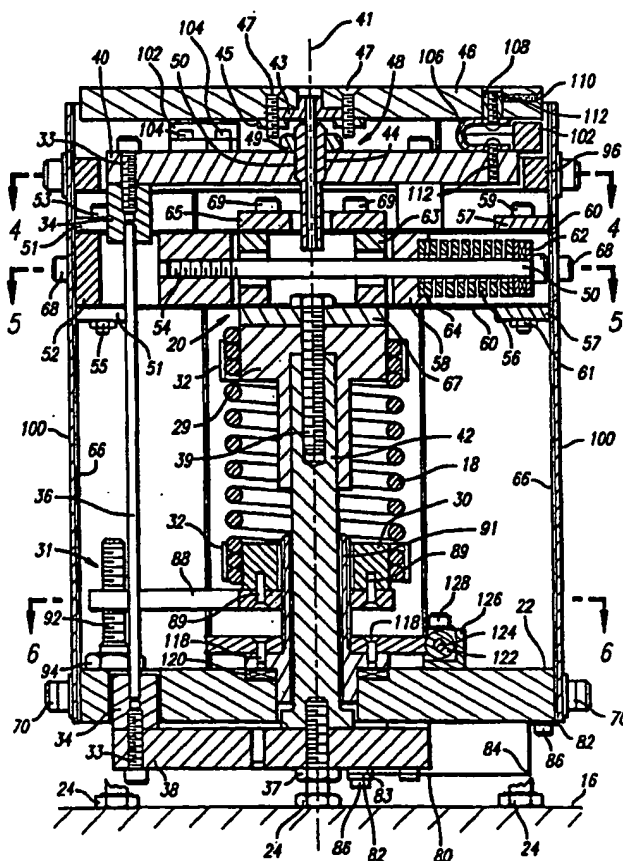
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(57) Abstract

An improved version of a vibration isolation system using negative stiffness incorporates a payload (15) and payload platform (14) on just one 6 DOF isolator (12) in a unique and innovatively compact configuration. The isolator (12) includes a platform (14) supported on an assembly of independently acting flexure mechanisms which are connected in a serial fashion, tilt on top of horizontal on top of vertical, and in turn connected to a base (12). Proper arrangement of the mechanisms and the payload/platform center of mass (17) provides highly effective decouple isolator performance. In addition, an innovative flexure preloading method which significantly improves vertical isolation performance is incorporated. This method can be used with prior (unsymmetric) designs or combined with a set of shear flexures (66) in an innovative symmetric arrangement to provide more assurance of ideal decouple response to mutually perpendicular base excitation input. A geared mechanism (116) is also incorporated to facilitate manual adjustment of a main support spring (18) whenever payload weight is changed.



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VIBRATION ISOLATION SYSTEM

This application is a continuation in part of co-pending application Serial No. 08/125,496 filed September 22, 1993 which is a continuation of application
5 Serial No. 07/987,046 filed December 4, 1992 entitled "VIBRATION ISOLATION SYSTEM", which is a continuation-in-part of application Serial No. 07/708,995, filed May 31, 1991, entitled "Vibration Isolation System", now U.S. Patent No. 5,178,357, which
10 is a continuation-in-part of application Serial No. 07/681,808, filed April 8, 1991, entitled "DAMPED VIBRATION ISOLATION SYSTEM", which is a continuation-in-part of application Serial No. 395,093, filed August 16, 1989, entitled "VIBRATION ISOLATION
15 SYSTEM".

BACKGROUND OF THE INVENTION

Field of the Invention:

This invention relates generally to suspension systems and methods for isolating and reducing the
20 transmission of vibratory motion between an object and a base and, more particularly, to a compact omnidirectional vibration isolation or suspension system that exhibits low stiffness, high damping to limit resonant responses of the system, effective isolation at the
25 higher frequencies, high isolator resonant frequencies, and can accommodate changing weight loads without significantly degrading isolation system performance. This invention also relates more specifically to a class of suspension systems which provides six-degree-of-
30 freedom isolation (translation and rotation in all directions) in which the isolator system is a single compact assembly centered underneath the platform in pedestal fashion, rather than an array of three or four isolators spread out around the periphery of the
35 platform.

Description of Related Art:

The problems caused by unwanted vibration on equipment, devices and processes that are extremely motion sensitive have been widely researched and numerous solutions to prevent or reduce the transmission of vibratory motion have been proposed and developed. Many of the devices designed to reduce the transmission of unwanted vibration between an object and its surroundings, commonly called vibration isolators or suspension devices, have utilized various combinations of elements such as resilient pads made from a variety of materials, various types of mechanical springs, and pneumatic devices. There are, however, shortcomings and disadvantages associated with these particular prior art isolation systems which prevent them from obtaining low system natural frequencies and from limiting resonant responses to low values while providing high isolation performance at the higher frequencies.

These shortcomings and disadvantages of prior art systems were addressed through the development of the novel vibration isolation systems and novel devices and methods for retro-fitting existing vibration isolation systems described in application Serial No. 07/395,093, filed August 16, 1989, entitled "VIBRATION ISOLATION SYSTEM", application Serial No. 07/681,808 filed April 8, 1991, entitled "DAMPED VIBRATION ISOLATION SYSTEM", application, Serial No. 708,995, filed May 31, 1991, entitled "VIBRATION ISOLATION SYSTEM," and application Serial No. 08/125,496 filed September 22, 1993 entitled "VIBRATION ISOLATION SYSTEM", which are all hereby incorporated by reference in this present application. The particular vibration isolation systems described in these applications and utilized in connection with the present invention provides versatile vibration isolation by exhibiting low stiffness in an axial direction (generally the direction of the payload weight) and any direction substantially transverse to the axial direction

(generally a horizontal direction), as well as tilt or rotation about three mutually perpendicular axes. The particular system utilizes a combination of uni-directional or bi-directional isolator subassemblies that can be connected together in series fashion to provide omni-directional isolation. Each isolator is designed to isolate the axial or the transverse component of any vibratory translation to effectively isolate vibrations along or about any directional axes. In subsequent discussions, an axial-motion isolator will be referred to as a vertical-motion isolator, and the system of axial-motion isolators will be referred to as the vertical-motion isolation system. Similarly, a transverse-motion isolator will be referred to as a horizontal-motion isolator, and the system of transverse-motion isolators will be referred to as the horizontal-motion isolation system. Lastly, a tilt-motion isolator in conjunction with a mechanism allowing rotation about an axis parallel to the tilt axes will be referred to as the tilt-motion isolation system.

In the embodiments described in the above-noted applications, the isolator relies on a particular principle of loading a particular elastic structure which forms the isolator or a portion of it (the loading being applied by either the supported weight or by an external loading mechanism) to approach the elastic structure's point of elastic instability. This loading to approach the point of elastic instability, also called the "critical buckling load" of the structure, causes a substantial reduction of either the vertical or the horizontal stiffness of the isolator to create an isolation system that has low stiffness in the vertical and in any horizontal direction, and increases the damping inherent in the structure. While stiffness is reduced, these isolators still retain the ability to support the payload weight.

If the load on an elastic structure with an instability is greater than the critical buckling load, the excessive load will tend to propel the structure into its buckled shape, creating a "negative-stiffness" or
5 "negative-spring-rate" mechanism. By combining a negative-stiffness mechanism with a spring, adjusted so that the negative stiffness cancels or nearly cancels the positive stiffness of the spring, one obtains a device that can be placed at or near its point of elastic
10 instability. The magnitude of the load causing the negative stiffness can be adjusted, creating an isolator that can be "fine tuned" to the particular stiffness desired.

SUMMARY OF THE INVENTION

15 The present invention provides a method to achieve mass properties of a combined payload and payload platform which, when supported by one of the 6-DOF isolators described in my co-pending application (Serial No. 08/125,496) or as described herein with added novel
20 and significant improvements, results in an isolation system that is unusually compact and simple without compromising any performance features. The method provides for adjustment of platform mass properties and tilt flexure stiffness so that the platform can
25 accommodate passively a small amount of redistribution in payload weight, as occurs for example in systems with moving stages. A novel and effective method has been devised to add small, discrete damping elements to control resonant amplitudes, while still providing nearly
30 ideal damped high frequency isolation. Unique and significant improvements have also been added to the isolation mechanism with a radial flexure preload system which uses an internal tension member and soft spring that is self-contained rather than requiring a separate
35 external load path to "ground". This preload method can be used in any of my prior designs, or in a novel symmetric arrangement of vertical plate flexures

described herein that eliminates a potential source of vertical and lateral motion coupling. This preload method greatly simplifies the radial flexure preloading and fine tuning processes. The present invention also includes an improvement by superimposing a gear wheel on the vertical load adjustment screw, such that it can be driven by a shaft-mounted worm gear. The shaft extends to the exterior of the isolator to better facilitate manual adjustment.

10 Briefly, the present invention is directed to a single isolator platform, particularly the arrangement and functioning of the isolator components which makes effective omni-directional vibration isolation possible in a single isolator design. The present invention is an extension of the isolator described in my co-pending application (Serial No. 08/125,496), with the addition of a platform and payload whose combined center-of-mass (CM) is made to coincide or nearly coincide with the vertical, horizontal and rotational force center of the isolator.

20 The vertical and horizontal force centers of the isolator are those points at which small vertical or horizontal forces, respectively, applied to the isolator cause only vertical or horizontal translation, respectively, without rotation. The rotational force center is the point at which a small moment applied to the isolator causes rotation only and no translation. This method works with any 6-DOF isolator that possesses a common point through which the net vertical, horizontal and rotational reaction forces act. This point in the present invention is the mid-point of the tilt flexure. Placement of the CM at or near the mid-point of the tilt flexure provides this essential feature in the present invention.

The present invention utilizes a platform with mass properties such that, when combined with the payload mass properties, results in a combined CM that is located at or very near the midpoint of the tilt flexure. The tilt flexure is supported on, and is operationally in series

with the horizontal-motion isolator which is supported on, and is operationally in series with the vertical-motion isolator which is supported on the base or the vibrating foundation. Small shifts in the payload CM, as
5 might occur for example during stepper stage motion, are accommodated by a tilt flexure with sufficient stiffness to resist any static imbalance, along with sufficient rotational inertia to limit attendant angular acceleration. Small loose weights can be placed on the
10 platform for initial leveling. Leveling centers the payload platform with respect to the tilt stops for equal tilt accommodation in all directions, and keeps the CM directly above the tilt flexure to avoid tilt vertical-motion coupling. Vertical location of the CM with
15 respect to the tilt flexure is important to minimize or effectively eliminate tilt-lateral motion coupling.

One embodiment of the present invention incorporates an innovative and simpler means of applying and finely adjusting preload on the radial flexures. This preload
20 provides negative stiffness in the vertical direction. Preloading is achieved in the present invention by merely turning (tightening) a loading spring adjustment screw after the unit is aligned and assembled.

In the present invention, tensioning means--that is,
25 a tension bolt--passes right through a widely clearanced hole made in the center hub and operationally attaches to the far ends of the opposing radial flexures. This device does not require a separate load path to "ground" as represented by the loading cantilevers and adjustment
30 cantilevers in previously described embodiments. Instead, it "floats" along with the center hub and radial flexure subassembly. Between one end of the tension bolt and the corresponding outer end of one set of radial flexures is a soft die spring which provides a higher
35 degree of compliance and more constant preload force than was possible with the adjustment and loading cantilevers, thus allowing longer travel of the vertical isolator.

The more travel that can be built into an isolator, the larger the displacement disturbance that can be accommodated; thus, the better the performance. At its other end the tension bolt threads into a block, such
5 that setting or adjusting radial flexure preloading--both coarse and fine--is achieved by turning this tension member into or out of the block.

For this new preloading method to work, the support structure at the outer end of the radial flexures on at
10 least one side must be very compliant relative to the die spring stiffness because, upon deflection of the center hub, the preload will actually be reduced in accordance with the stiffness of that outer support point and the degree to which the radial flexures shorten. In the
15 present invention, this compliance is provided by a single thin, but wide, "sheet metal" flexure. The width provides needed stiffness in the direction perpendicular the radial flexures, and the reduced thickness provides compliance in the radial direction.

20 Another embodiment of the present invention incorporates elastomeric damping elements into the vibration isolation system in order to control transmissibility at the system natural frequencies. The elastomeric material exhibits the desired, nearly
25 classical visco-elastic damping effect. The elements are attached to parts of the isolator that move relative to one another, where the relative motion represents a degree of freedom to be isolated. Each degree of freedom can be selectively damped. A symmetric arrangement of
30 elements is normally used for each damped degree of freedom. Various shapes--such as cylinders or U-shaped channels--and thicknesses are used to achieve specific amounts of damping for each degree of freedom. For example, damping only for the tilt-motion can be provided
35 by a set of three U-shaped elements placed between the payload plate and the column top plate, each at an equal distance from the center of the isolator and at an

angular spacing of one hundred and twenty (120) degrees. The isolator has simple interfacing features designed and built in so that damping elements can be easily retro-fitted, removed or replaced.

5 The present invention improves performance by eliminating a source of coupling between the vertical and lateral motions. The vertical-motion mechanism must be stiffly restrained laterally to maintain stability of the mechanism and to avoid low internal isolator resonances.

10 The center hub and radial flexure subassembly is restrained on all four sides in a balanced fashion by vertical "plate" type or sheet metal flexures which restrain the center hub against lateral motion by in-plane shear resistance. The thin and wide sheet metal

15 flexure supports the outer end of the radial flexures on both sides of the center hub, not just one. Another pair of sheet metal flexures are arranged in a similar fashion in the same plane but perpendicular to the radial flexures, and are attached to the center hub by a

20 corresponding pair of very thin and wide (radial) plate flexures. Thus, lateral motion of the center hub is prevented by the same type of mechanism in both (mutually perpendicular) directions. The plate flexures are inherently very stiff against in-plane shear deformation.

25 A worm gear subassembly has been added for an improved means of load adjustment. During installation and set-up, the vertical "null" position of the real operating payload must get positioned to midpoint of the vertical stroke. This is accomplished with a load

30 adjustment screw that raises or lowers the base of the main vertical load carrying spring. The adjustment screw is at the center of the isolator, and in present invention the screw has a gear-tooth outer circumference that meshes with a worm gear mounted on a shaft extending

35 out through the cover of the isolator assembly. The exposed end of the shaft is readily turned by hand.

BRIEF DESCRIPTION OF THE DRAWINGS

FIGURE 1 is an elevation view of one embodiment of a vibration isolation system made in accordance with the present invention showing an isolator, a weighted platform and a payload;

FIGURE 2 is a plan view of the isolation system shown in Figure 1;

FIGURE 3 is a partial cross-sectional side view of the isolator shown in Figure 1 taken along line 3--3;

FIGURE 4 is a partial cross-sectional plan view of the isolator shown in Figure 1 taken along line 4--4 in Figure 3;

FIGURE 5 is a partial cross-sectional plan view of the isolator shown in Figure 1 taken along line 5--5 in Figure 3;

FIGURE 6 is a partial cross-sectional plan view of the isolator shown in Figure 1 taken along line 6--6 in Figure 3.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

As shown in the drawings for purposes of illustration, the present invention is embodied in a more compact vibration-isolating suspension system comprising a payload supported on a weighted platform which is in turn supported on a single vibration isolator. Vertical-motion isolators made in accordance with the present invention are capable of larger vertical deflections with less stiffening due to nonlinear effects, easier adjustment of vertical stiffness and possible improved performance by elimination of a possible source of vertical-motion and horizontal-motion coupling.

Figure 1 shows one embodiment of an isolation system made in accordance with the present invention. The system 10 uses a single composite 6-DOF isolator 12 that supports a weighted platform 14 that in turn supports a

payload 15 with center-of-mass (CM) 17. The isolator 12 and platform 14 support the payload 15 relative to a foundation 16 and reduce the transmission of omnidirectional vibrations between the payload 15 and foundation 16.

Referring now to Figures 1 and 2, one embodiment of a weighted platform is illustrated. The platform 14 includes an upper plate 19 that is connected to a weight ring 21 by four tubes 23 and threaded rods 25 that screw into the plate 19 and hold the plate 19, the ring 21 and tubes 23 together with nuts 25. The weight ring 21 lowers the CM of the platform 14 so that the combined CM of the platform 14 and payload 15 with CM 17 is at or very near the center of the tilt flexure of the isolator 12, which corresponds to the vertical, horizontal and rotational center of the isolator. With this configuration tilt-horizontal-motion coupling and tilt-vertical-motion coupling is eliminated or nearly eliminated. Location of the CM of the payload directly or nearly directly over the tilt flexure can be accomplished by moving the payload 15 on the platform 14, or by moving loose weights (not shown) on the platform. This also levels the platform 14 to keep it floating off tilt stops in the isolator.

Referring now to Figures 1, 3, 4, 5 and 6, one embodiment of an isolator made in accordance with the present invention is illustrated. As in the composite isolator described in my co-pending application, Serial No. 08/125,496, the isolator comprises a vertical-motion isolator operatively connected vertically in series with a horizontal-motion isolator which is operatively connected vertically in series with a tilt-motion isolator. The vertical-motion, horizontal-motion and tilt-motion isolators are arranged in the same general configuration as in that co-pending application.

The vertical-motion isolator of the present invention includes a support spring 18 in the form of a

coil spring that operatively connects between a center hub assembly 20 and a base plate 22 that is connected to the foundation 16 by three leveling screws 24. The center hub assembly 20 comprises a center block 63, a center clamp 65, a center plate 67 and the upper spring support 29, clamped together with screws 69 and 39. A negative-stiffness mechanism also connects to the center hub assembly 20 to cancel stiffness from the support spring 18, and includes radially-compressed radial flexures 60. Clamps 32 secure the support spring 18 to the upper spring support 29 and to a lower spring support 30, and an anti-rotation device 31. Improvements in accordance with the present invention include the means for supporting and loading the radial flexures 60, the means for constraining horizontal motions of the center hub assembly 20 and the means for manually raising and lowering the lower spring support 30 to accommodate changes in weight load. These will be described later.

The horizontal-motion isolator comprises three flexible beam-columns 36 (only one of which is shown in Figure 3) in the form of thin cylindrical rods that are operatively connected between a lower column plate 38 and an upper column plate 40. The beam-columns 36 are press-fit into end fittings 34 which are attached to the upper column plate 40 and to the lower column plate 38 with screws 33. The lower column plate 38 connects to a stiff support shaft 42 with screw 37. The support shaft 42 is connected with screw 39 to the upper spring support 29 and to the center plate 67 which are part of the center hub assembly 20, thereby providing a relatively rigid connection between the top of the vertical-motion isolator and the bottom of the horizontal-motion isolator. The upper column plate 40 which is the top of the horizontal-motion isolator connects to a top mounting plate 46 through a tilt flexure assembly 48 which acts as the tilt-motion isolator. The tilt flexure assembly 48 comprises a tilt flexure 41 which is a thin

cylindrical rod press-fit into end fittings 43 and 44 which are secured to the top mounting plate 46 and upper column plate 40. End fitting 41 is secured to the top mounting plate 46 with tilt flexure retainer 45 and screws 47. End fitting 46 is threaded at its outside surface 50 and is screwed into the upper column plate 40 and secured with lock nut 49.

An isolator cover comprises four cover panels 100 and four corner panels 98 that attach to the base plate 22 with screws 70. A stop plate 96 also attaches to the cover panels 100 and corner panels 96 with screws 70 and serves as the down-motion stop with three motion stops 102 spaced 120 degrees and attached to upper column plate 40 with screws 104. The motion stops 102 also serve as tilt-motion stops to limit tilt of the top mounting plate 46 with respect to the upper column plate 40.

Three tilt dampers 106 spaced at 120 degrees comprise U-shaped strips of viscoelastic damping material attached between the upper column plate 40 and the top mounting plate 46. Screws 112 attach the lower ends of the dampers 106 to the upper column plate 40 and the upper ends of the dampers 106 to bushings 108 secured in holes in the top mounting plate with set screws 110.

The four radial flexures 60 are clamped to load block 52, the center hub assembly 20 and spring block 58 and are compressed using a tension screw 50 and die spring 56. The flexures 60 are clamped to load block 52 using clamps 51, with bolts 53 and nuts 55 and they are clamped to spring block 58 using clamps 57 with bolts 59 and nuts 61. They are clamped to the center hub between clamp 65 and plate 67 and the center block 63 with screws 69. The threaded end 54 of tension screw 50 screws into load block 52 and the head of screw 50 bears against a thrust bearing 62 which bears against die spring 56 which is recessed into cavity 64 in spring block 58. Preloading the radial flexures 60 and fine tuning the load to adjust the negative-stiffness effect are both

accomplished by turning screw 50. The blocks 52 and 58 are attached to vertical thin-sheet flexures 66 by screws 68. The flexures 66 are attached at their lower ends to base plate 22 using screws 70.

5 This method of supporting and compressing the radial flexures 60 provides a constant compressive force on the flexures 60 by permitting a softer spring means to load the radial flexures 60 and a softer support structure in the radial direction. As the center hub 20 deflects
10 vertically, the radial flexures 60 shorten in the radial direction thereby reducing the compressive load and the negative-stiffness effect and increasing the vertical stiffness of the isolator and the vertical natural frequency of the system. The softer the elastic
15 structure providing the compressive load, the more constant the compressive load as the center hub deflects away from its equilibrium center position and the smaller the increase in vertical stiffness and vertical natural frequency. In the present configuration, the stiffness
20 of the die spring 56 and the flexures 66 affect the reduction in compressive load as the radial flexures 60 shorten since these elements act in parallel. The flexures 66 are very flexible in the radial direction parallel to the radial flexures but very stiff in the
25 horizontal direction perpendicular to the radial flexures. This method of support constrains the center hub against deflections in directions perpendicular to the radial flexures 60. Two additional thin-sheet shear flexures 72 are oriented horizontally and are clamped
30 under the lower radial flexures 60 to the center hub assembly 20 and to blocks 76 with screws 79. The blocks 76 are attached to vertical thin-sheet flexures 74 with screws 78. The flexures 74 are attached near their bottom ends to the edge of base plate 22. The shear
35 flexures 72 are stiff in directions parallel to the radial flexures 60 and constrain the center hub 20 from displacements in this direction. They are very flexible

in bending and, with the high flexibility of the flexures 74 in a horizontal direction perpendicular to the shear flexures 72 they add little vertical stiffness to the isolator. Thus, with the system of flexures described
5 that support the radial flexures 60 and connect to the center hub assembly 20 and base plate 22, the center hub assembly 20 is constrained with high stiffness against movement in the horizontal plane.

Three bottom flexures 80 (Figure 3 and hidden lines
10 in Figure 6) spaced 120 degrees apart and operatively connected between the lower column plate 38 and the base plate 22 provide a stiff constraint for the lower column plate 22 and the lower end of support shaft 42 against horizontal motion. These flexures with the other
15 flexures in the system provide a stiff constraint for the center hub assembly 20 against all motions of the mass associated with the center hub assembly 20, the support shaft 42 and lower column plate 38, relative to the base plate 22, thereby producing high internal structural
20 resonances of the isolator structure associated with these modes of deformation. These modes can represent the lowest isolator structural resonant frequencies.

The anti-rotation device 31 prevents torsional deformation of support spring 18 as the lower spring
25 support 30 is raised and lowered to accommodate changes in weight load on the isolator. The anti-rotation device 31 comprises a plate 88 that is clamped to the lower spring support 30 with screws 89 and has a U-shaped slot 90 that engages a stop post 92 that is screwed into a
30 hole in base plate 22 and secured with lock nut 94. The plate 88 has a threaded hole that engages a threaded support tube 91 that is supported on a thrust bearing 120 that is contained in a recess in base plate 22. A worm gear 116 is attached to support tube 91 with screws 118
35 and engages worm 122 that is mounted on adjusting screw/shaft 124 that runs on bearings in housing 126 that is attached to base plate 22 with screws 128. The worm

is secured to adjusting screw/shaft 124 with pin 130. Rotation of screw/shaft 124 raises or lowers the lower spring support 30 to accommodate changes in weight load on the isolator.

5 A novel feature of the present invention as illustrated in Figures 1-6 is the use of a weighted platform on a single 6-DOF isolator made in accordance with the present invention to provide a compact and efficient isolated platform for reducing the transmission
10 of vibrations between an object and a base. An important feature of the isolator that permits its use in this manner results from its vertical, horizontal and rotational force centers all coinciding at a point corresponding to the mid-point of the tilt flexure 41
15 near the top of the isolator. This feature results from the use of a vertical-motion isolator operatively in series with a horizontal-motion isolator which is operatively in series with a tilt-motion isolator. By using a weighted platform with its CM below the isolator
20 force center to balance an object on the platform with its CM above the isolator force center, the combined CM of the platform, object and ballast weights, if needed, can be located at or near the isolator force center to decouple or nearly decouple the vertical, horizontal and
25 rocking input vibrations to give optimum or near-optimum isolation performance. Having the isolator force center at the tilt flexure 41 near the top of the isolator permits a compact weighted platform that accommodates an object with a reasonably high CM relative to its base.
30 Locating the CM of the object, or object and ballast if needed, above or nearly above the tilt flexure 41 can be achieved by moving the object 15 and/or ballast weights (not shown in the figures) on the platform 14 in order to float the top mounting plate 46 off the tilt stops 102
35 and to provide what leveling is needed. Locating the CM of the total suspended weight vertically at or near the isolator force center can also be achieved with the aid

of ballast weights and/or vertical spacers for the payload or object supported on the platform. The tubes 23 that locate the weight ring 21 relative to the platform upper plate 19 can also be varied in length. It is not necessary to have the CM of the total suspended weight right at or very close to the isolator force center. A significant amount of coupling between the vertical, horizontal and rocking motions can be tolerated and the system can still provide effective isolation.

10 The tilt flexure 41 can be stiffened as necessary to readily facilitate floating the top mounting plate 46 off the tilt stops 102 and leveling the mounting plate. This increases the rocking natural frequencies of the system but very low vertical and horizontal natural frequencies can be maintained. Higher rocking frequencies can usually be tolerated because rocking vibration inputs are typically very small compared with vertical and horizontal vibration inputs. Tilt damping can also serve to limit the resonant rocking responses.

20 Another novel feature of the present invention is the use of a soft spring to load the radial flexures 60 and provide a more constant radial load and negative-stiffness effect for the vertical-motion isolator, along with the novel use of a system of flexures to provide a stiff restraint against motion of the center hub and associated mass to maintain high isolator structural resonant frequencies. This feature allows a simpler means of preloading and fine-tuning the compressive load on the radial flexures 60, and provides a more constant vertical stiffness during vertical deflection of the isolator due to less reduction in radial load associated with shortening of the radial flexures. The particular isolation system configuration and construction methods illustrated in Figures 1-6 is just one of many arrangements which can incorporate the novel features of the present invention. Other configurations and construction methods will become apparent to those

skilled in the art. For example, instead of supporting the load block 52, and thereby the left ends of the radial flexures 60, on thin-sheet flexures 66, this load block 52 could be attached to a stiff plate structure that connects to the base plate 22 similar to the construction in my co-pending application (Serial No. 08/125,496). The shear flexures 72 could then be eliminated since the center hub assembly 20 would be restrained in the radial direction by the radial stiffness of the radial flexures 60 and the stiff plate structure. There would be some coupling between vertical and horizontal motions of the isolator as the center hub is pulled toward the stiff plate structure from shortening of the radial flexures from vertical displacement of the center hub, but this effect would be very small.

The elements making up the present structure can be made from common structural materials such as steel and aluminum alloys. Other structural materials having suitable strength, elastic and mass properties can also be used.

While one particular form of the invention has been illustrated and described, it will be apparent that various modifications can be made without departing from the spirit and scope of the invention. Accordingly, it is not intended that the invention be limited, except by the attached claims.

WHAT IS CLAIMED IS:

1. An omni-directional vibration isolation suspension system for supporting an object in an equilibrium position relative to a base while suppressing transmission of vibratory motion between the object and
5 the base, comprising:

a vertical-motion isolator for isolating vibratory motion in a vertical direction operatively connected in series with a horizontal-motion isolator for isolating vibratory motion in the horizontal direction;

10 a tilt motion isolator for isolating rotational motion operatively connected in series with said vertical-motion isolator and horizontal motion isolator, said tilt motion isolator supporting a platform upon which the object rests, said platform having a center of
15 mass below the center of mass of the object such that the combined center mass of the platform and object is located relative to said vertical-motion isolator, horizontal-motion isolator and tilt motion isolator to substantially decouple the vertical, horizontal and
20 rotational input motions on the system.

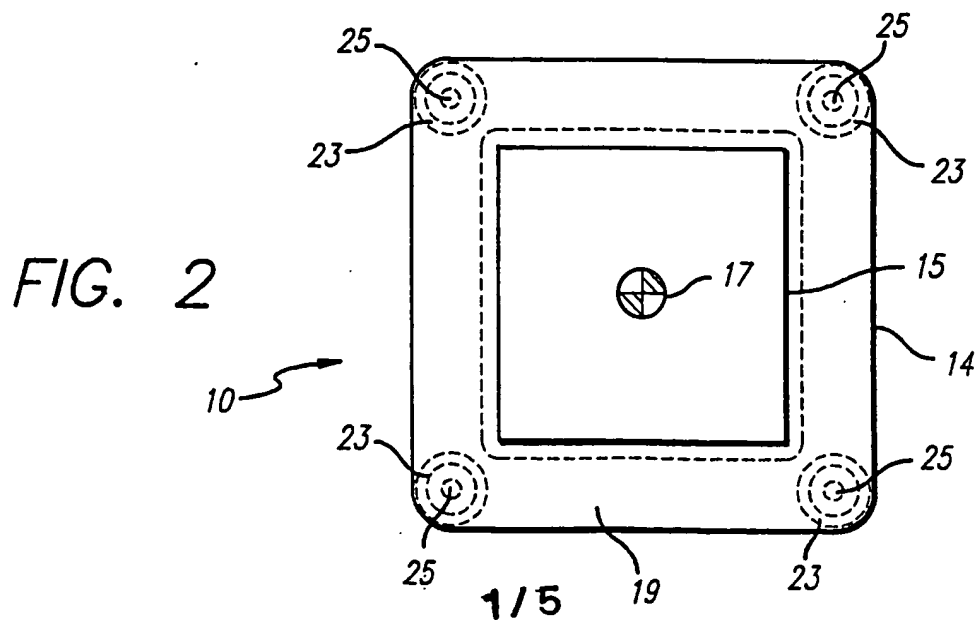
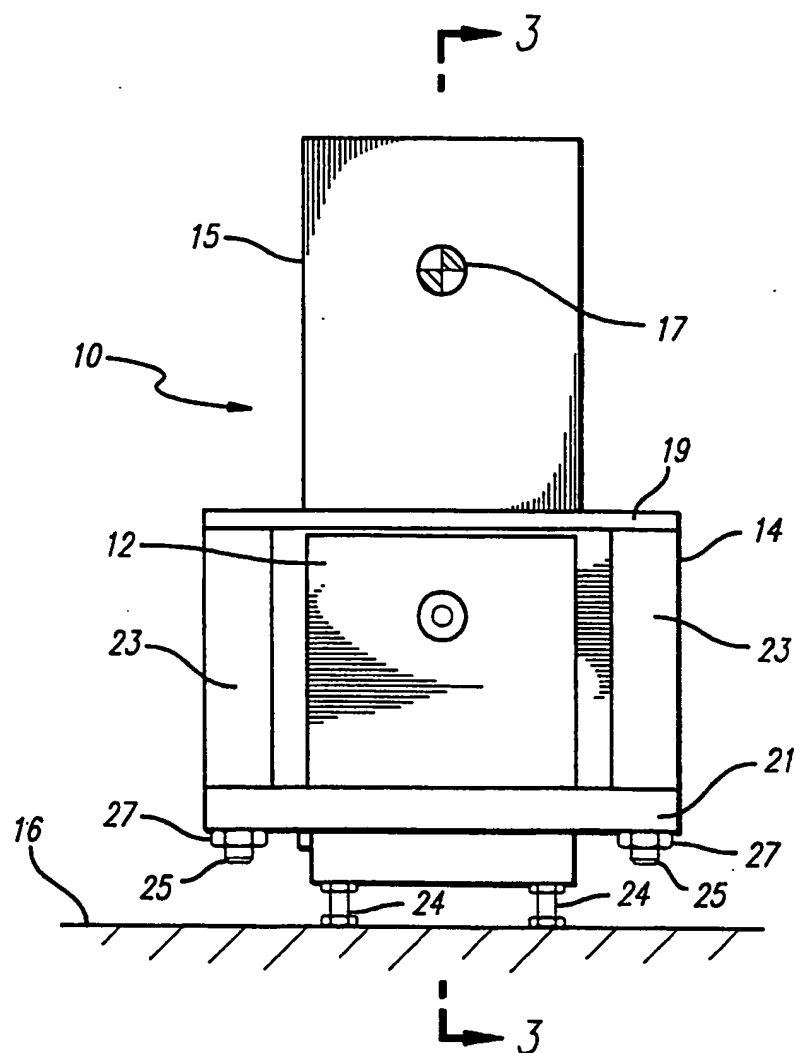
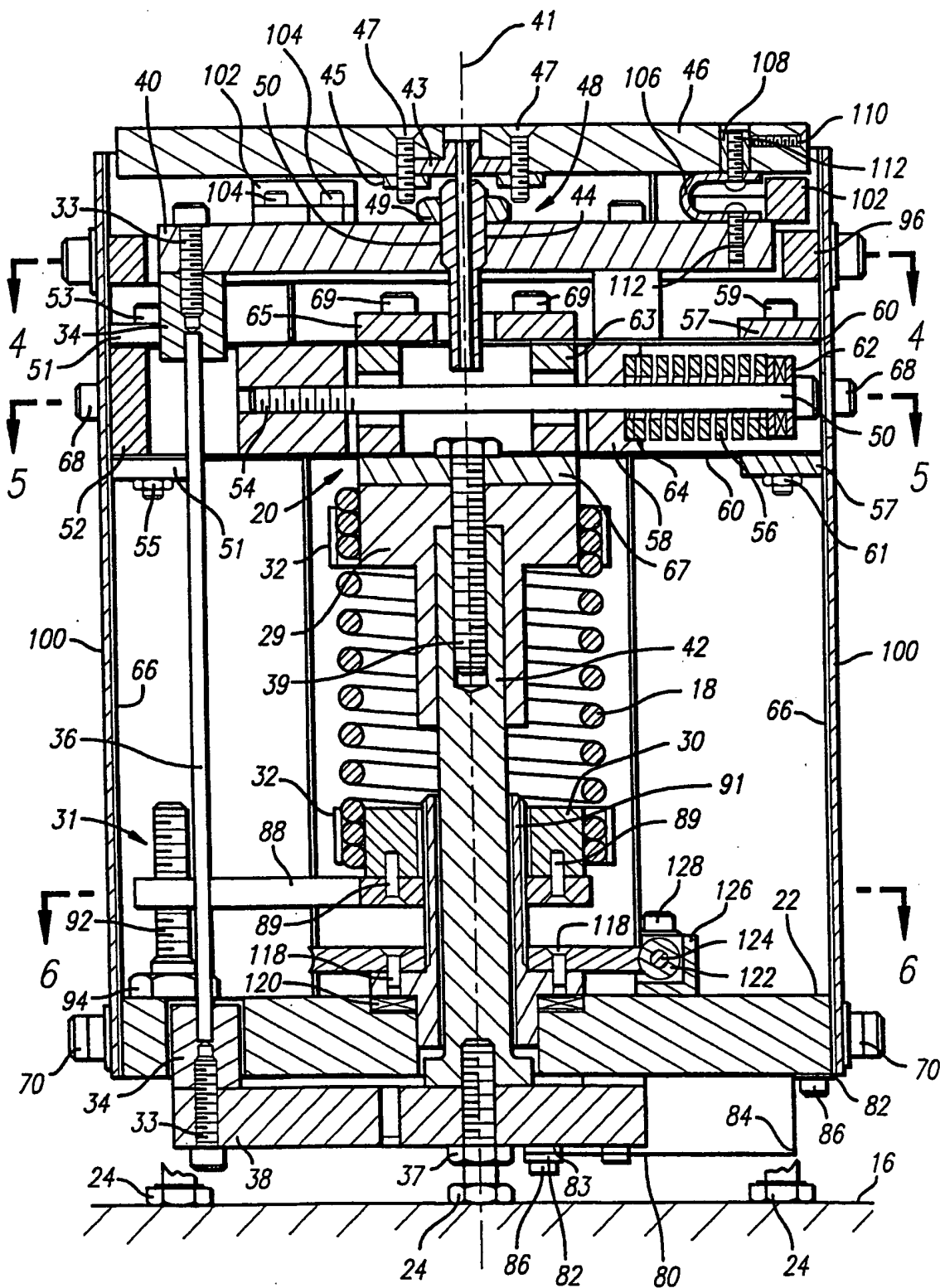


FIG. 3



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SUBSTITUTE SHEET (RULE 26)

FIG. 4

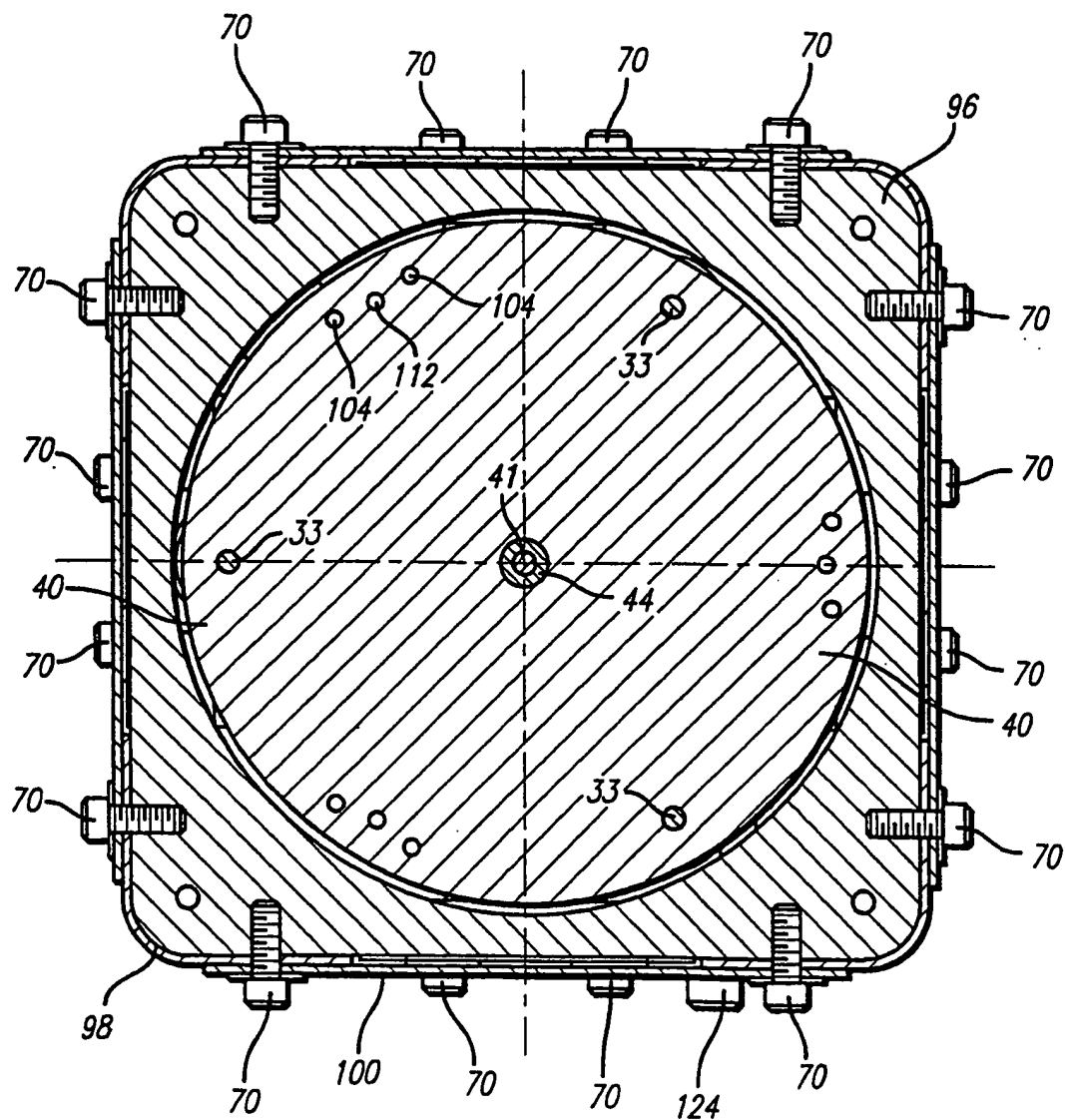


FIG. 5

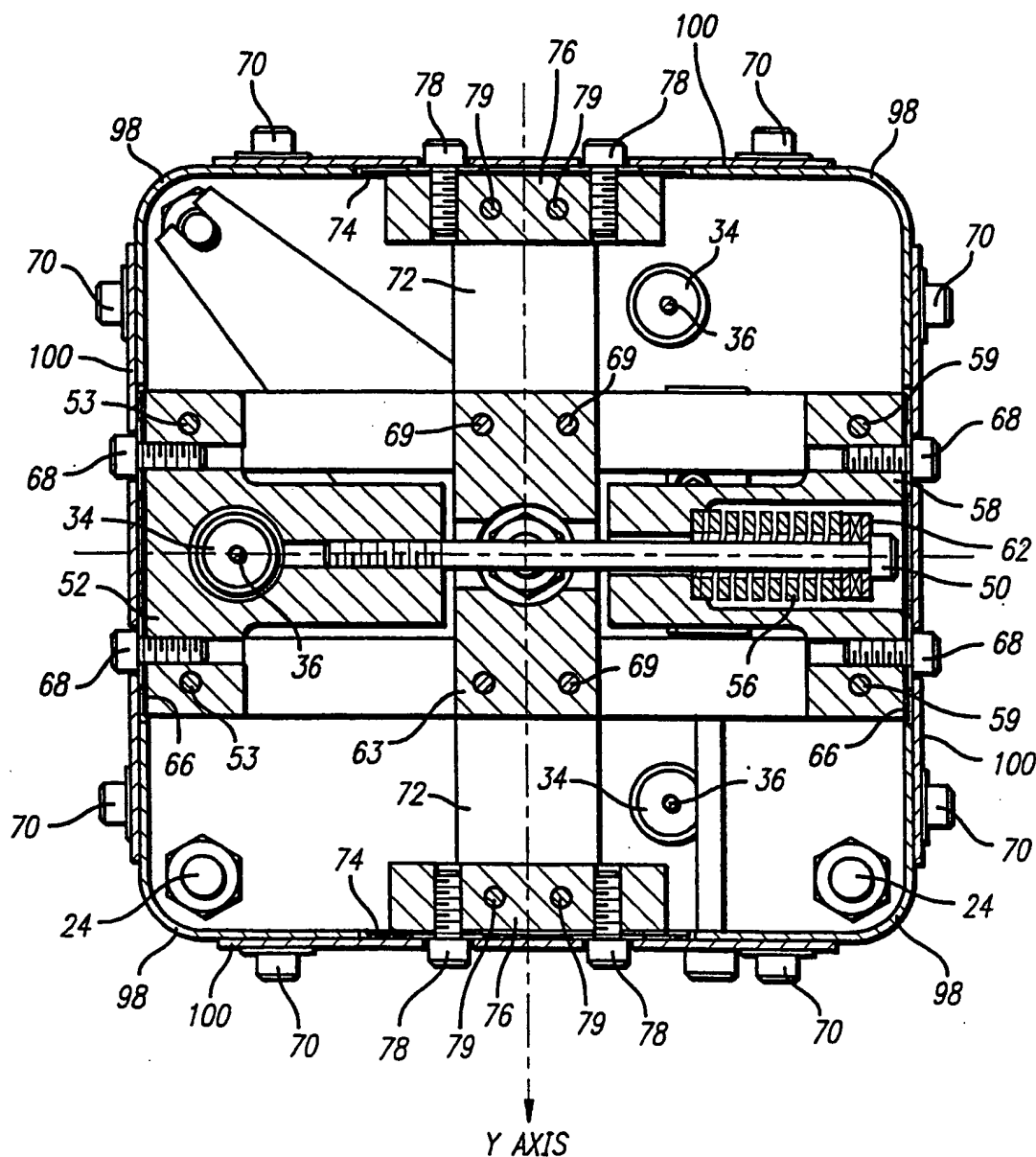
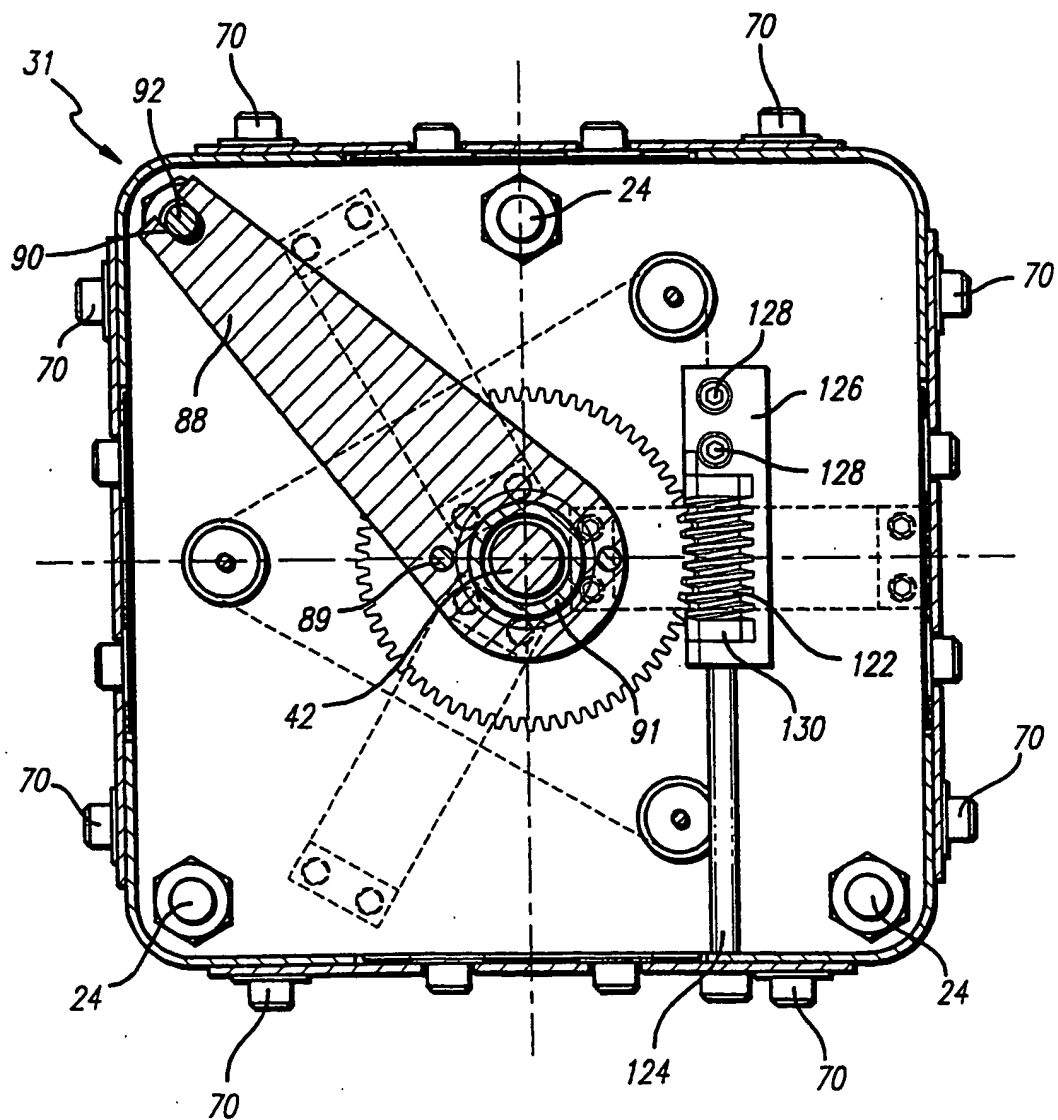


FIG. 6



INTERNATIONAL SEARCH REPORT

International application No.
PCT/US95/00907

A. CLASSIFICATION OF SUBJECT MATTER

IPC(6) : F16F 15/04

US CL : 248/619

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

U.S. : 248/619, 620, 621, 623, 603, 632, 638, 678

267/140.5, 36.1

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US,A 4,871,142 (deMey, II) 03 October 1989, see whole document	1



Further documents are listed in the continuation of Box C.



See patent family annex.

* Special categories of cited documents:

A document defining the general state of the art which is not considered to be of particular relevance

E earlier document published on or after the international filing date

I document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

O document referring to an oral disclosure, use, exhibition or other means

P document published prior to the international filing date but later than the priority date claimed

T

later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

X

document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

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document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

Z

document member of the same patent family

Date of the actual completion of the international search

09 MARCH 1995

Date of mailing of the international search report

24 MAR 1995

Name and mailing address of the ISA/US
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